Purdue University

Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1994

Dynamic Simulation of Rotor Contact Forces in Twin Screw Compressors

G. P. Adams
University of Arkansas

W. Soedel
Purdue University

DYNAMIC SIMULATION OF ROTOR CONTACT FORCES IN TWIN SCREW COMPRESSORS

G.P. Adams
University of Arkansas
Mechanical Engineering Department
Fayetteville, AR 72701

Werner Soedel
Purdue University
1077 Herrick Laboratories
West Lafayette, IN 47907

ABSTRACT

A model is presented for computing the rotor contact forces in twin screw compressors. The objective is to incorporate the contact forces, along with the compression loads, in determining the bearing forces. The model allows only a rotational degree of freedom for each rotor. A backlash type clearance is modelled between contact points in the rotor meshing zone. This results in a piecewise linear model in which two separate system modes are possible. Separate systems of equations of motion are used to represent each mode. In addition, methods for computing the motion due to transitions between the states are also presented. The system of equations is solved using a Runge-Kutta integration algorithm with time as the independent variable and the rotor angular positions and velocities as the dependent variables. For each rotor, the moment about the rotor axis due to the compression process is input as a function of the rotor angular position. The simulation is implemented for a typical set of compressor parameters and the resulting bearing forces are computed. The results indicate that the compression loads, as compared to the contact force, dominate the bearing forces.

OVERVIEW OF MODEL

This model is based on research conducted by the authors while at Purdue University [1]. An overview of the model is presented here.

Basic Assumptions Utilized

The following basic assumptions are applied in developing the model of the rotor interaction.

1. The rotors are assumed to be rigid bodies. The flexibility of the male lobes and female flutes is neglected.
2. The rotor shafts are assumed to be rigid, with no torsional or lateral flexibility.
3. The bearing mounts are assumed to be rigid, with no lateral displacements.

These assumptions result in a 2-degree-of-freedom model, each rotor having only a single rotational degree of freedom about its central axis. The state of the system is defined by the angular position and velocity of each of the rotors.

Rotor Contact

The contact between the rotors is modelled using points on the rotor pitch circles, as seen in Figure 1. The pitch circles define the kinematic relationship between the rotors. The radii of the pitch circles are assumed to remain constant. A backlash type of clearance is modelled by allowing only one contact point, C, to exist on the male rotor pitch circle, while two contact points, A and B, exist on the female pitch circle. The clearance region is then specified as the angle between points A and B. The *contact* mode is defined when point C is in contact with either point A or B. The *independent* mode is defined when point C is in the clearance region.

Load Parameters

The compression loads on the rotors must be included in the model. A method for computing the compression loads was developed by the authors [2]. Each component of the compression loads is required to compute the bearing forces. However, for the model described above, only the moment load about the rotor axis is required. These loads are represented in the model as a function of the male angular position.

The compressor is modelled as a direct drive type with the male rotor shaft as the input. During normal operation, instantaneous variations in the motor torque occur, due to changes in the compression loads. The relationship between instantaneous torque and rotational speed is assumed to be linear. The parameters describing this relationship are included in the differential equations of motion for the rotors.

Damping loads are included in the model. Some amount of rotational damping from the bearings is inherent in the physical system. In addition, the damping can be used to aid the numerical evaluation by damping the transients associated with the initial conditions.

Equations of Motion

One set of coupled differential equations defines the rotor motion during the contact mode. A separate set of independent differential equations defines the motion during the independent mode. The transition between modes is defined by the impact model and the loss of contact conditions.

Equations of Motion for the Contact Mode

Figure 1 is a schematic diagram of the rotors representing contact between the male lobes and the female flutes. Euler's equations are applied to each of the rotors. This results in coupled equations for the angular acceleration of each rotor.

$$I_m\ddot{\theta}_m = T_I - M_{mz} - c_m\dot{\theta}_m - Nr_m \tag{1}$$

$$I_f\ddot{\theta}_f = -M_{fz} - c_f\dot{\theta}_f + Nr_f \tag{2}$$

where

$$\begin{aligned}
\dot{\theta}_{m/f},\, \ddot{\theta}_{m/f} &= \text{angular velocity/acceleration of the male/female rotor} \\
r_{m/f} &= \text{pitch radii of the male/female rotor} \\
c_{m/f} &= \text{damping coefficient applied to the male/female rotor} \\
M_{mz/mf} &= \text{moment about the } z \text{ axis of the male/female rotor due to compression} \\
I_{m/f} &= \text{moment of inertia of the male/female rotor} \\
N &= \text{contact force between the male and female rotors} \\
T_I &= \text{electromotor input torque.}
\end{aligned}$$

The kinematic constraints, $\dot{\theta}_f = \frac{r_m}{r_f}\dot{\theta}_m$ and $\ddot{\theta}_f = \frac{r_m}{r_f}\ddot{\theta}_m$ provide the coupling between the equations of motion.

Solving each of the equations for the contact force and equating the results, one can obtain a relationship for the angular acceleration of the male rotor in terms of the angular acceleration and velocity of the female rotor. The kinematic constraints can then be applied to obtain an equation for the angular acceleration of the male rotor in terms of the applied forces, the angular velocity of the male rotor and the physical parameters of the two rotors. The input torque can be modelled as, $T_I = T_0 - \Delta T \dot{\theta}_m$. The resulting equation for $\ddot{\theta}_m$ is

$$\ddot{\theta}_m = \frac{\left(\frac{T_0 - M_{mz}}{r_m}\right) - \left(\frac{M_{fz}}{r_f}\right) - \dot{\theta}_m \left(\frac{1}{r_m} c_m + \frac{r_m}{r_f^2} c_f + \frac{\Delta T}{r_m}\right)}{\frac{1}{r_m} I_m + \frac{r_m}{r_f^2} I_f}. \tag{3}$$

Once the angular acceleration and velocity of the male rotor are computed, the kinematic constraints can be used to compute the angular velocity and acceleration of the female rotor.

Equations of Motion for the Independent Mode

When the rotors are not in contact the kinematic constraints no longer apply and the equations of motion are not coupled. These equations can be integrated separately.

$$\ddot{\theta}_m = \frac{T_0 - M_{mz} - (\Delta T + c_m)\dot{\theta}_m}{I_m} \tag{4}$$

$$\ddot{\theta}_f = \frac{-M_{fz} - c_f \dot{\theta}_f}{I_f}. \tag{5}$$

Determination of Transitions

A transition defines a change in the contact mode. The basic approach is to determine the final conditions which result from the end of the current mode. These are then used as the initial conditions for the next mode.

Transition from Contact to Independent Modes

The first transition to consider is that from the contact mode to the independent mode. The contact force between the rotors must be compressive, as a tensile force is not physically possible. This is the key to defining the loss of contact between the rotors.

The loss of contact can be determined by solving for the contact force, using the current values of angular acceleration and angular velocity. If the solution results in a tensile contact force then the true solution requires that the rotors are no longer in contact.

When a transition from the contact mode to the independent mode occurs, the system state at the transition time is used as the initial conditions for the independent system of differential equations.

Transition from Independent to Contact Modes, Impact

The transition from the independent mode to the contact mode involves an impact situation. This transition is detected by monitoring the system state during the independent mode. An impending impact is defined when the relative position of the contact points is within a specified tolerance and the relative velocity displays that the points are approaching one another.

The impact model used here consists of applying the impulse momentum equations along with a coefficient of restitution, e, to give an equation for the angular velocity of the male rotor after impact. Subscripts 1 and 2 denote before and after impact, respectively.

$$\dot{\theta}_{m_2} = \frac{\left(I_m - e\left(\frac{r_m}{r_f}\right)^2 I_f\right)\dot{\theta}_{m_1} + (1+e)\left(\frac{r_m}{r_f}\right) I_f\,\dot{\theta}_{f_1}}{I_m + \left(\frac{r_m}{r_f}\right)^2 I_f} \tag{6}$$

The angular velocity of the female can then be expressed in terms of the angular veloctiy of the male.

$$\dot{\theta}_{f_2} = \frac{e\left(r_m\dot{\theta}_{m_1} - r_f\dot{\theta}_{f_1}\right) + r_m\dot{\theta}_{m_2}}{r_f}. \tag{7}$$

Once these computations are completed, the impulse, $F_{ave}\Delta t$, can be obtained from the change in angular momentum

$$F_{ave}\Delta t = \left(\frac{I_m}{r_m}\right)\left(\dot{\theta}_{m_2} - \dot{\theta}_{m_1}\right). \tag{8}$$

The impact is assumed to occur instantaneously. Therefore, the change in velocities occurs without a change in position. The system state after impact is used as the initial conditions for the subsequent contact mode.

CONCLUSIONS

The model described here has been implemented to simulate several typical operating conditions. The associated bearing forces were computed based on the configuration shown in Figure 2. The bearing force F_{x1_m}, for a typical operating condition is shown in Figure 3. In this figure, the contributions due to the contact force and the compression loads are shown separately. It is evident that the compression loads dominate the resulting bearing forces. A frequency plot of the total bearing force is shown in Figure 4. Due to the sharpness of the compression loads, a rich harmonic content is evident.

ACKNOWLEDGMENTS

The authors would like to acknowledge United Technologies, Carrier Corporation for sponsoring the research program which made these efforts possible.

REFERENCES

[1] G. P. Adams. *Modelling and Computer Simulation of Rotor Chatter and Oscillating Bearing Loads in Twin Screw Compressors.* PhD thesis, Purdue University, Purdue University W. Lafayette, IN 47907, December 1993.

[2] G.P. Adams and Werner Soedel. A method for computing the compression loads in twin screw compressors. In Woerner Soedel, editor, *1994 International Compressor Engineering Conference at Purdue*, Herrick Laboratories, West Lafayette, IN, 47907, July 1994. Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University.

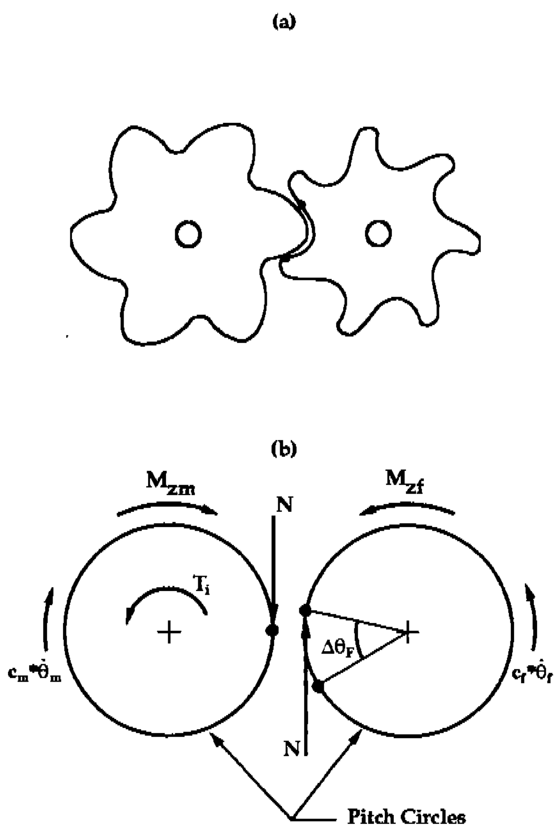

Figure 1: (a) Clearance between rotors, (b) schematic diagram of rotorcontact model.

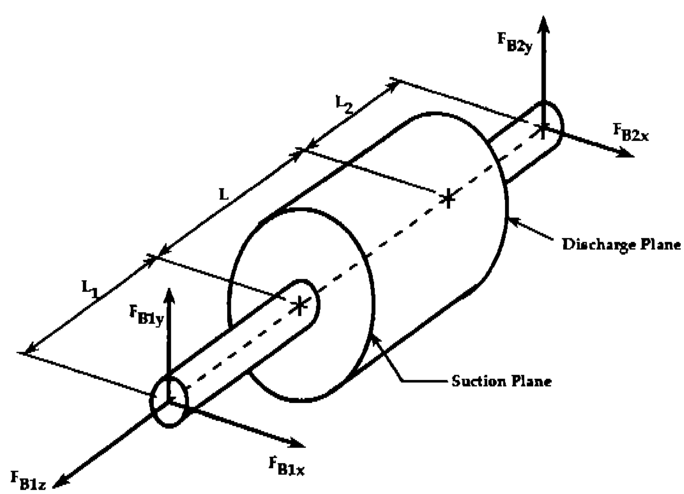

Figure 2: Typical bearing configurations for a single compressor rotor.

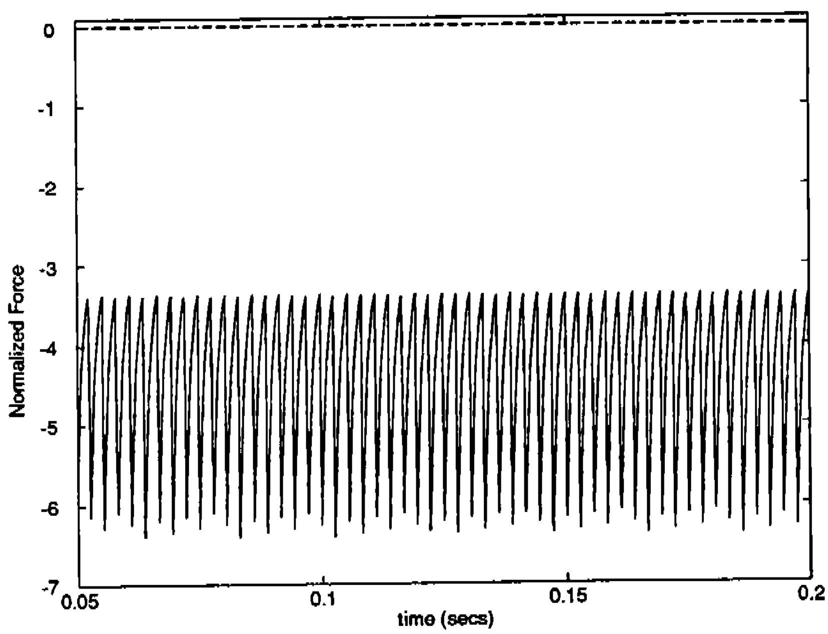

Figure 3: Male rotor suction bearing force, F_{B1x}, (under-pressure);
(– – –) force due to rotor contact, (—) force due to compression loads.

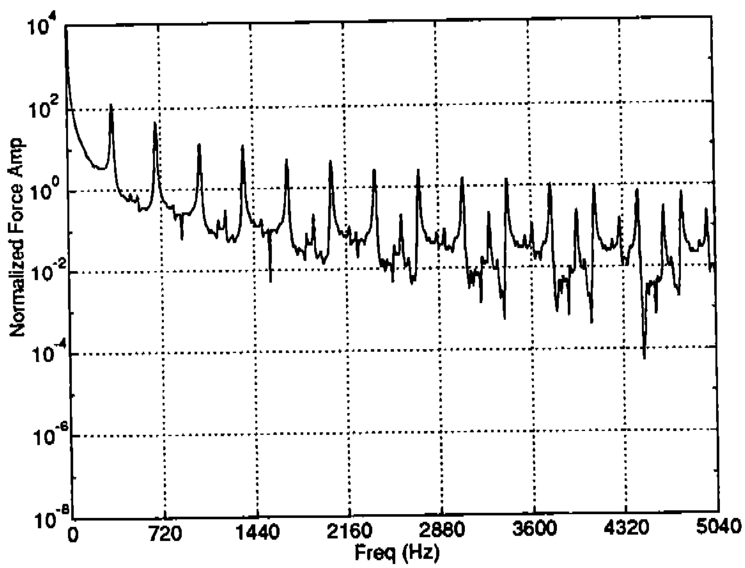

Figure 4: Frequency spectra, F_{B1x} (under-pressure);.